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EMPIRICAL ASSESMENT OF THE MAIN DRIVING SYSTEM OF THE CIRCULAR SAWING MACHINE

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Abstract
The producers of panel saws tend to improve sawing accuracy and minimise a level of vibrations, to increase their competitiveness at the market. Mechanical vibrations in the main saw driving system, which level depend on a plethora independent factors, may really affect sawing accuracy and general machine tool vibrations. The objective of the research was to explore vibrations signals of the main spindle system, and to extract informative features for assessment of the current design of the panel saw Fx3. Accelerations of the main spindle system body in X, Y and Z axes of the panel saw co-ordinate system were measured with the vibration tester Fluke 810. The both the root-mean-square value and the peak amplitude value of vibrational velocities of the body of the main spindle of the panel saw allowed us to assess vibrational quality of the examined machine tool. From this point of view the analysed system might be classified in the quality range between "rather good" and "good". These results confirmed that in the design of the panel saw potential quality possibilities are still hidden in reserve.

Key words: *panel saw, spindle, vibration velocity, vibration tester, vibrational quality*

INTRODUCTION

Several different sawing processes can be employed by a sawmill either in the primary breakdown process or in secondary breakdown. Usually, the effect of sawing is not perfect, and abnormal occurrences in the sawing processes can cause deviations from the ideal shape. In case of lumber, there are distinguished six types of lumber shapes caused by sawing problems: thin snake, fat snake, snipe, flare, taper and wedge. Although there are several ways in which machine-caused lumber shape defects can reduce the profitability of a sawmill, the economic impact of these defects on the industry is unknown (Rasmussen et al. 2004). Among machine tools used in sawmills, circular sawing machines with a vertical or an horizontal arbor (spindle) seem to be most applied. A tool (circular saw blade, band saw blade or saw blade) is a weakest element (most flexible) in the machine tool structure. For that reason in the literature there are several works devoted to tool behavior in static (Wang Houli and Siekkinen 1983; Wasielewski and Orlowski 2007) and dynamic conditions either during idling (Mohammadpanah and Hutton 2015a; Sandak et al. 2007; Wasielewski et al. 2009) and cutting (Mohammadpanah and Hutton 2015b; Orlowski and Wasielewski 2006; Potočnik et al. 2013).

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On the other hand, vibrations caused by the kinematics of the machine tool such as a sash gang saw (frame sawing machine), with a crank mechanism in the main driving system, can affect the occupational environment and an operator, who is exposed to the high vibration level over a long period of time (Goglia and Grbac 2005). Impinging on the occupational environment of the sawmill in the case of frame sawing machines could be minimized if in the driving system a design proposed by Wasielewski and Orlowski (2002) is applied. However, even in this case the operator is still under the influence of alternating cutting forces. The phenomenon of vibration coupling between bandsaw frame and feed carriage system has been examined by Okai (2009). In case of chain saws Operational Deflection Shapes (ODS), which are vibration patterns of a structure when it is forced to vibrate under particular stationary operating conditions, have been studied by Kromulski at al. (2010). Empirical research into dynamics of sawing process with chain saws have been also carried out by Axelsson (1968), and Wójcik and Skarżyński (2008).
In the furniture industry panel saws are commonly used. For the latter machine tools errors such as snaking of the longitudinal kerf, rough sawn surfaces or even a washboard pattern may occur. In panel saws saw blades are usually clamped with collars (Mohammadpanah and Hutton 2015b; Svoreň and Hrčková 2015). It should be emphasised that the quality effects depend on dynamical properties of the whole structure of the machine tool (machine tool – workpiece – clamping device – tool). Hu Wan-yi et al. (2003) reported findings of the work devoted to noise generated by the panel saw when unload running, and on the other hand, Kvietková et al. (2015) published results of the effect of number of saw blade teeth on noise level during transverse cutting of beech wood. Generally, the panel saws produces two kinds of noise: airflow noise (aeromechanic noise) and structural noise. Mechanical vibration noise could come from unbalance of the main saw blade system, the main spindle eccentric, loose of bearing and incorrect assembly (Hu Wan-yi et al. 2003). Noise is accompanied by vibration and they do not exist separately, hence, if the noise is decreased simultaneously vibrations are at lower level. Moreover, mechanical vibrations in the main saw driving system, which level depend on a plethora independent factors, may really affect sawing accuracy and general machine tool vibrations. The producers of panel saws tend to improve sawing accuracy and minimise a level of vibrations, to increase their competitiveness at the market. The objective of the research was to explore vibrations signals of the main spindle system, and to extract informative features for assessment of the current design of the panel saw Fx3.

THEORETICAL BACKGROUND

According to Dietrych (1978), the ability to realise the tasks (objective function) can be assessed by the indices of quality to which can be included expected life, reliability, precision and low level of emitted disturbances (vibrations and noise). These indices are inseparably connected with the character of the dynamic process and also of the vibroacoustical processes taking place in machine tools (Cempel 2009). The casual relationship between the quality indices and vibrations are presented in Fig. 1. It could be emphasised that each quality index may be obtained if in the machine tool small vibrations are present.
Vibration diagnostic investigations can be carried out generally with a very simple set of measuring equipment comprised the following: vibration meter with possibility of measuring either acceleration, or velocity or displacement; a portable instrument for collect data; and a portable spectrum analyser software for field and laboratory analyses.

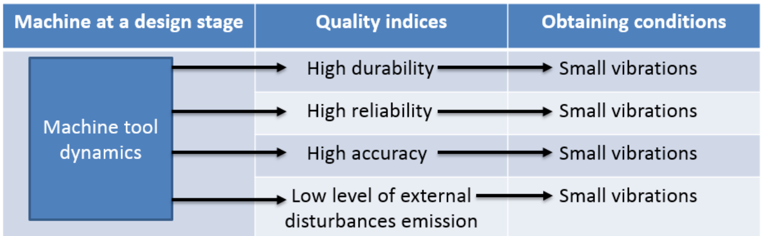

Figure 1. Indices of machine tool quality and their connections with vibrations (Cempel 2009)

According to Cempel (2009) determination of the standard of diagnosis is risky, it is better to limit diagnosis to one definite industrial case, with repeated methodology. Conducting of vibroacoustic empirical tests the researcher ought to answer himself for a question: Is it always worth measuring the vibration velocity and not displacement or acceleration? (Cempel 2009).

MATERIALS AND METHODS

Vibration diagnostic investigations has been carried out on the main spindle system of the panel saw Fx3 (f. REMA SA, PL). The circular saw blade ∅300×3.2×30, z = 96 was mounted on the spindle and spindle rotational speed (measured) was equal to 5128 rpm. The measurement point was located on top of the main spindle system body (Fig. 2a), in which accelerometer A was placed. Accelerations in *X, Y, Z* axes were measured with the vibration tester Fluke 810 (f. Fluke, USA). In the plant conditions obtained results might be analysed on a display of the tester or later in the laboratory with the use of Fluke Viewer Software on PC.

a) b)

Figure 2. General view of the spindle system of the panel saw Fx3 with a position of the accelerometer *A* (a) and a vibration tester Fluke 810 (b), where: *X, Y, Z* – axes of a the panel saw co-ordination system

RESULTS ANALYSES

In Figure 3a waveform of the acceleration signal, obtained from the vibration tester Fluke 810, in Z axis direction is presented, from which an average value of acceleration or acceleration peak-to-peak value can be estimated.

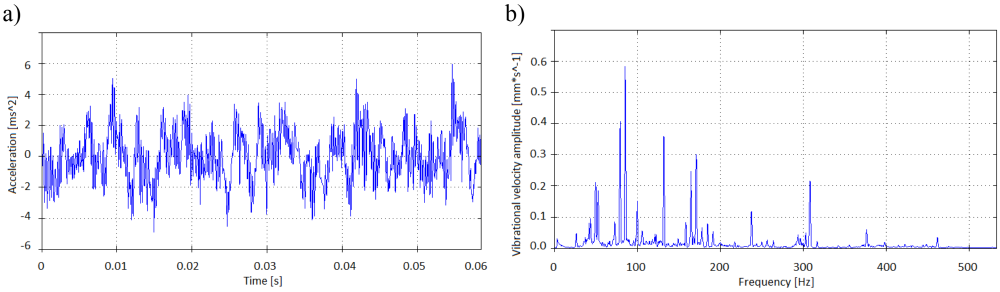

Figure 3. Waveform of the acceleration signal (a) and vibrational velocity spectra of the signal of the panel saw Fx3 in the measurement point (Fig. 2a) in *Z* axis

The vibrational velocity spectra of the main spindle system in *Z* axis direction is shown in Fig. 2b. On the basis of vibrational velocity amplitudes it is possible to compute changes of component velocities v_j versus time t from the equation:

$$v_j(t) = A_{vj}(f_j)\sin(2\pi f_j t) \tag{1}$$

where: $A_{vj}(f_j)$ is a vibrational velocity amplitude for frequency f_j. If equation (1) is integrated we can obtain an equation for component of displacement versus time:

$$D_j(t) = -\frac{A_{vj}(f_j)}{2\pi f_j}\cos(2\pi f_j t) \tag{2}$$

Total changes of vibrational velocities or total displacement may be obtained if a sum of signal components is made, in both cases total values are presented in Fig. 4 as Sigma courses. The Vibration calculator (AMES 2016) gives an another way of calculation displacements and acceleration.

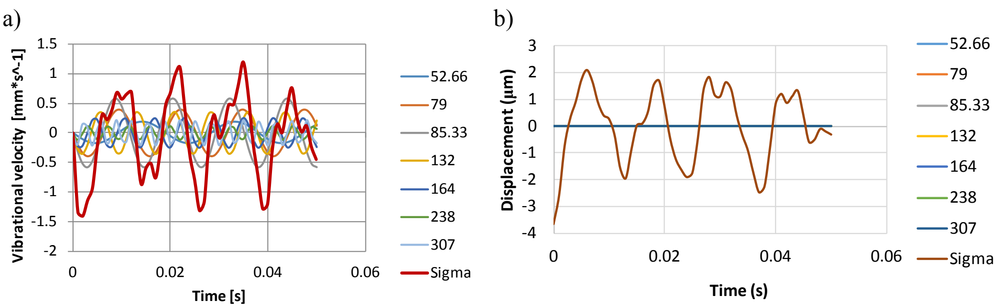

Figure 4. Plots of displacement velocities (a) and total displacements (b) versus time of the panel saw Fx3 at the measurement point (Fig. 2a) in *Z* axis direction

From the plots of vibrational velocities for each measurement axis of the panel saw Fx3, *RMS* (root-mean-square) values of vibrational velocities were calculated, and then total vibrational velocity for the measurement point was determined from the equation as follows:

$$v(RMS) = \sqrt{v_X(RMS)^2 + v_Y(RMS)^2 + v_Z(RMS)^2} \qquad (2)$$

The results of calculations are shown in Fig. 5. Since, in the literature there is the confusion of the criterion values (Cempel 2009) two following values were determined: the root-mean-square amplitude of the total vibrational velocity and the peak value (amplitude). The RMS value of the total vibrational velocity is equal to 2.08 $\mathrm{mm \cdot s^{-1}}$, whereas the peak value in this measurement point equals 3.27 $\mathrm{mm \cdot s^{-1}}$. According to Blake's guidelines the peak amplitude is in the range of admissible values (Cempel 2009). On the other hand Łączkowski (Cempel 2009) defined a range of the peak velocity for machine tool equal to 2.5-6.3 $\mathrm{mm \cdot s^{-1}}$. Moreover, an American diagnostic company IRD Machanalysis (Cempel 2009) published own standards, and according to them the root-mean-square value is in the range between "good" and "admissible". Furthermore, if recommendations of the British company VCI are taken into account, from the point of view of vibrational limit condition (the obtained root-mean-square value of vibrational velocity) it can be said that the examined main spindle system of the panel saw Fx3 might be classified between "rather good" and "good".

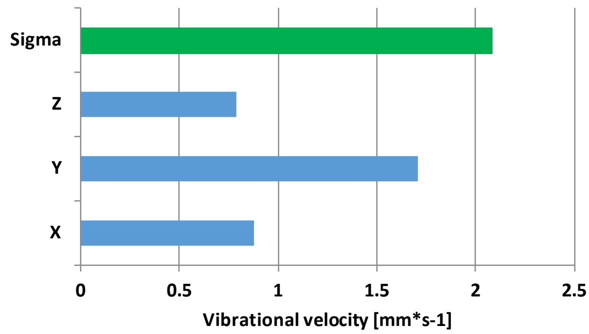

Figure 5. Root-mean-square values of vibrational velocities in directions of *X, Y* and *Z* axes of the panel saw co-ordination system measured at the measurement point (Fig. 2a) together with a resultant *RMS* of vibrational velocity Sigma

CONCLUSIONS

On the basis of the obtained results during carried out experiments, and conducted analyses of both the root-mean-square value and the peak amplitude value of vibrational velocities of the body of the main spindle of the panel saw Fx3 it can be stated that from the vibrational point of view the analysed system might be classified in the quality range between "rather good" and "good". These results confirmed that in the design of the panel saw potential quality possibilities are still hidden in reserve.

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